

**UNITED STATES PATENT APPLICATION FOR:**

**RADIAL PUMPING OIL SEAL FOR A FLUID DYNAMIC BEARING MOTOR**

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## **RADIAL PUMPING OIL SEAL FOR FLUID DYNAMIC BEARING MOTOR**

### **CROSS-REFERENCE TO RELATED APPLICATIONS**

[0001] This new application for letters patent claims priority from an earlier filed provisional patent application entitled "Radial Pumping Oil Seal For Fluid Dynamic Bearing Motor." That application was filed on May 7, 2003 and was assigned Application No. 60/468,803.

### **BACKGROUND OF THE INVENTION**

#### **Field of the Invention**

[0002] The present invention relates to fluid dynamic bearing motors. More specifically, the present invention pertains to fluid dynamic bearing motors such as are used to support and rotationally drive one or more memory discs.

#### **Description of the Related Art**

[0003] The computer industry employs magnetic discs for the purpose of storing information. This information may be stored and later retrieved using a disc drive system. Computer systems employ disc drive systems for transferring and storing large amounts of data between magnetic discs and the host computer. The magnetic discs are typically circular in shape (though other shapes are known), and are comprised of concentric, or sometimes spiraled, memory tracks. Each track contains magnetic data. Transitions in the magnetic data are sensed by a magnetic transducer known as a read/write head. The transducer is part of the disc drive system, and moves radially over the surface of the disc to read and/or write magnetic data.

[0004] **Figure 1** presents a perspective view of magnetic media **10** as are commonly employed for information storage. In this view, a plurality of stacked magnetic discs **10'** is shown. The discs **10'** in **Figure 1** are shown in vertical alignment as is common within a disc drive system. Each disc **10** has a central concentric opening **5** for receiving a spindle (shown at **51** in **Figure 2**). A rotary

motor drives the spindle **51**, causing the discs **10** of the disc pack **10'** to rotate in unison.

[0005] As noted, the disc **10** itself is supported on a drive spindle **51**. The drive spindle **51** rotates the disc **10** relative to the magnetic head assembly **58**. **Figure 2** provides a perspective view of a disc drive assembly **50**. In this arrangement, a plurality of discs **10'** are stacked vertically within the assembly **50**, permitting additional data to be stored, read and written. The drive spindle **51** receives the central openings **5** of the respective discs **10**. Separate suspension arms **56** and corresponding magnetic head assemblies **58** reside above each of the discs **10**. The assembly **50** includes a cover **30** and an intermediate seal **32** for providing an air-tight system. The seal **32** and cover **30** are shown exploded away from the disc stack **10'** for clarity.

[0006] In operation, the discs **10** are rotated at high speeds about an axis (not shown). As the discs **10** rotate, the air bearing slider on the head **58** causes the magnetic head **58** to be suspended relative to the rotating disc **10**. The flying height of the magnetic head assembly **58** above the disc **10** is a function of the speed of rotation of the disc **10**, the aerodynamic lift properties of the slider along the magnetic head assembly **58** and, in some arrangements, a biasing spring tension in the suspension arm **56**.

[0007] The servo spindle **52** pivots about pivot axis **40**. As the servo spindle **52** pivots, the magnetic head assembly **58** mounted at the tip of its suspension arm **56** swings through arc **42**. This pivoting motion allows the magnetic head **58** to change track positions on the disc **10**. The ability of the magnetic head **58** to move along the surface of the disc **10** allows it to read data residing in tracks along the magnetic layer of the disc. Each read/write head **58** generates or senses electromagnetic fields or magnetic encodings in the tracks of the magnetic disc as areas of magnetic flux. The presence or absence of flux reversals in the electromagnetic fields represents the data stored on the disc.

[0008] In order to accomplish the needed rotation of discs, an electric motor is provided. The electric motor is commonly referred to as a "spindle motor" by virtue of the drive spindle **51**, or "hub," that closely receives the central opening **5** of a disc **10**. **Figure 3** illustrates the basic elements of a known spindle motor design, in cross-section. The motor **400** first comprises a hub **410**. The hub **410** includes an outer radial shoulder **412** for receiving a disc (not shown in **FIG. 3**). The hub **410** also includes an inner shaft **414**. In this arrangement, the shaft **414** resides and rotates on a stable counterplate **440**. A sleeve **420** is provided along the outer diameter of the shaft **414** to provide lateral support to the shaft **414** while it is rotated.

[0009] It can be seen that a bearing surface **422**, or "journal surface," is formed between the shaft **410** and the surrounding sleeve **420**. In early arrangements, one or more ball bearing systems (not shown) was incorporated into the hub **410** to aid in rotation. Typically, one of the bearings would be located near the top of the shaft, and the other near the bottom. A raceway would be formed in either the shaft or the sleeve for holding the plurality of ball bearings. The bearings, in turn, would be lubricated by grease or oil. However, various shortcomings were realized from the mechanical bearing system, particularly as the dimensions of the spindle motor and the disc tracks became smaller. In this respect, mechanical bearings are not always scaleable to smaller dimensions. More significantly, in some conditions ball bearings generate unwanted vibrations in the motor assembly, causing the read/write head to become misaligned over the tracks. Still further, there is potential for leakage of grease or oil into the atmosphere of the disc drive, or outgassing of the components into this atmosphere.

[0010] In response to these problems, hydrodynamic bearing spindle systems have been developed. In these types of systems, lubricating fluid is placed along bearing surfaces defined around the rotating spindle/hub. The fluid may be in the form of gas, such as air. Air is popular because it avoids the potential for outgassing of contaminants into the sealed area of the head disc housing. However, air cannot provide the lubricating qualities of oil or the load capacity. Further, its low viscosity

requires smaller bearing gaps and, therefore, higher tolerance standards to achieve similar dynamic performance. As an alternative, fluid in liquid form has been used. Examples include oil and ferro-magnetic fluids. A drawback to the use of liquid is that the liquid lubricant should be sealed within the bearing to avoid leakage. Any loss in fluid volume results in a reduced bearing load capacity and life for the motor. In this respect, the physical surfaces of the spindle and of the housing would come into contact with one another, leading to accelerated wear and eventual failure of the bearing system.

[0011] Returning back to **Figure 3**, the motor **400** of **Figure 3** represents a hydrodynamic bearing system. A thrust plate **430** is disposed between the shaft **414** and the surrounding sleeve **420**. Fluid is injected in gaps maintained between the shaft **414** and surrounding parts, e.g., the counterplate **440**, the sleeve **420**, and the thrust plate **430**. The fluid defines a thin fluid film that cushions relative movement of hub parts.

[0012] The motor **400** is actuated by energizing coils in a stator in cooperation with one or more magnets. In the view of **Figure 3**, magnets **450** are seen disposed within the hub **410**, while stator coils **452** are provided on a base **460**. The magnets **450** and stator coils **452** interact to provide rotational movement of the hub **410**.

[0013] A means for retaining fluid within a hydrodynamically operated bearing surface for a spindle motor is to provide oil pumping grooves in the vertical journal bearing surface between the shaft and the sleeve or in the thrust bearing gap between the shaft and the counterplate. However, in the case of a straight-shaft journal bearing, axial space that could be used for journal bearing surface is rendered ineffective due to its being devoid of oil from the asymmetric pumping action of the seal. Also, since the voided area is not lubricated, bearing damage could result from contact in the non-lubricated area above the grooves during rotational excitement of the spindle. Thus, a need exists for an improved fluid dynamic bearing system for a spindle motor that retains liquid within and along the bearing surfaces. Further, there is a need for such a motor that reduces or

eliminates dry contact in the vertical journal bearing surface during rotation of the motor. Still further, there is a need for a hydrodynamic bearing arrangement that reduces the required length of the vertical journal bearing, as would be beneficial in the design of a hard drive for a lap-top computer, where such space is at a premium.

### **SUMMARY OF THE INVENTION**

[0014] The present invention provides an improved motor arrangement. The arrangement is useful in connection with rotary electrical motors, such as spindle motors in disc drive systems. More specifically, the invention is most applicable to motors that employ fluid dynamic bearing surfaces between relatively rotating parts.

[0015] In an exemplary arrangement, the improved spindle motor first comprises a hub having a shaft portion and an upper horizontal body portion. The motor also comprises a sleeve surrounding the shaft portion of the hub. A first fine vertical gap is retained between the shaft and the inner diameter of the surrounding sleeve. In addition, a fine horizontal gap is provided between the upper hub portion and the top of the sleeve. Optionally, a third fine gap is provided between an outer hub portion and the outer diameter of the sleeve. The first vertical gap is filled with a lubricating liquid, such as a clean oil. In one arrangement, the lubricating liquid extends into the horizontal gap and the third outer gap.

[0016] Preferably, a capillary seal is provided in the third fluid gap at one end. The capillary seal is disposed at an upper end of the third gap proximal to the second horizontal gap. In addition, oil pumping grooves are machined along the horizontal fluid gap. The oil pumping grooves may be machined into the top of the sleeve, though preferably they are machined into the bottom of the upper hub portion. The oil pumping grooves are used to impel oil towards the shaft of the hub. In this respect, the location of the oil pumping grooves prevents un-lubricated contact between the shaft and the sleeve and also requires a shorter sleeve and shaft than in conventional designs.

[0017] In an alternative embodiment, the improved spindle motor first comprises a hub fitted with a vertical sleeve portion inside of the hub. The motor also comprises a fixed shaft disposed within the sleeve. Fitted outside of the shaft over the sleeve is a shield. A first fine vertical gap is retained between the shaft and the inner diameter of the surrounding sleeve. In addition, a fine horizontal gap is provided between bottom of the shield and the top of the sleeve. Optionally, a third fine gap is provided between the shield and the outer diameter of the sleeve. The first vertical gap is filled with a lubricating liquid, such as a clean oil. In one arrangement, the lubricating liquid extends into the horizontal gap and the third outer gap.

[0018] Preferably, a capillary seal is provided in the third fluid gap at one end. The capillary seal is disposed at an upper end of the third gap proximal to the second horizontal gap. In addition, oil pumping grooves are machined along the horizontal fluid gap. The oil pumping grooves may be machined into the top of the sleeve, though preferably they are machined into the bottom of the shield. The oil pumping grooves are used to impel oil towards the shaft of the hub. In this respect, the location of the oil pumping grooves prevents un-lubricated contact between the shaft and the sleeve and also requires a shorter sleeve and shaft than in conventional designs.

#### **BRIEF DESCRIPTION OF THE DRAWINGS**

[0019] So that the manner in which the above recited features of the present invention can be understood in detail, a more particular description of the invention, briefly summarized above, may be had by reference to the appended drawings. It is to be noted, however, that the appended drawings (Figures 5-8) illustrate only typical embodiments of this invention and are therefore not to be considered limiting of its scope.

[0020] Figure 1 demonstrates a perspective view of magnetic media, i.e., thin film magnetic discs, as are commonly employed for information storage. In this view, a plurality of stacked discs is shown.

[0021] Figure 2 illustrates a perspective view of an exemplary disc drive assembly as might employ the improved spindle motor arrangement of the present invention.

[0022] Figure 3 provides a cross-sectional view of a known spindle motor arrangement.

[0023] Figure 4 presents a cross-sectional view of an improved spindle motor arrangement in which oil pumping grooves are machined.

[0024] Figure 5 illustrates an enlarged view of the gaps formed between the shaft of the spindle motor of Figure 5, and the surrounding sleeve.

[0025] Figure 6 depicts a bottom perspective view of the grooved portion of an exemplary hub in one embodiment.

[0026] Figure 7 presents a partial cross-sectional view of an alternate embodiment of an improved spindle motor arrangement in which oil pumping grooves are machined.

[0027] Figure 8 illustrates an enlarged view of the gaps formed between the shaft of the spindle motor of Figure 8, and the surrounding sleeve.

[0028] Figure 9 illustrates the results of a simulation exhibiting the pressure effect created by the oil pumping grooves during rotation of the shaft.

#### **DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENT**

[0029] The present invention provides an improved spindle motor arrangement 500, 600. The improved motor 500, 600 employs a novel radial pumping oil seal as a means of protecting the hub and sleeve from damage due to un-lubricated contact

between relatively moving components and a means to reduce the length of the shaft and sleeve.

[0030] **Figure 4** presents a cross-sectional view of an improved spindle motor arrangement **500** in one embodiment, in which oil pumping grooves **527** (see **Figure 6**) are machined. The motor **500** first comprises a hub **510**. The hub **510** includes a radial shoulder **512** for receiving and supporting a body to be rotated, such as a magnetic disc (not shown). The hub **510** includes a central shaft **514** and a horizontal body portion **518**. The shaft **514** is configured for constant high speed rotation. This rotation is established by a stator **552** which is mounted from a base **560**. The stator **552** typically defines an electric coil that, when energized, creates a magnetic field. The energized coil cooperates with magnets **550** which are mounted from an inner surface of the hub **510**.

[0031] As noted, the shaft **514** is configured for high speed rotation. In this respect, the shaft **514** rotates on a stationary counterplate **540**. **Figure 5** illustrates an enlarged view of the interfaces among the shaft **514**, counterplate **540**, sleeve **520**, and horizontal body portion **518** of the hub **510**. The interface between the bottom of the shaft **514** and the top of the counterplate **540** thus defines a thrust bearing **542**. Liquid lubricant is provided along the thrust bearing gap **542** to provide a fluid bearing surface. The top face of the counterplate **540** may optionally include a grooved pattern **544** for receiving and holding liquid lubricant when the motor **500** is at rest. When the motor **500** is at rest, the shaft **514** presses directly on the counterplate **540**. Fluid is then extruded around the outer diameter of the shaft **514**.

[0032] The motor **500** of **Figure 4** next comprises a sleeve **520**. In the arrangement of **Figure 5**, the sleeve **520** is stationary, and is supported on the counterplate **540**. The sleeve **520** is disposed between the rotating shaft **514** and the surrounding base **560**. Referring to **Figure 5**, it can be seen that the interface between the rotating shaft **514** and the surrounding sleeve **520** defines a second bearing surface **522**. The second bearing surface **522** is substantially vertical, and also receives liquid lubricant. This vertical bearing surface **522** can also be

configured with pumping grooves (not shown). Also notable is a third fluid-holding gap **572** located between the hub **510** and the outer diameter of the sleeve **520**. The third gap **572** may also receive fluid.

[0033] To prevent the shaft **514** and connected hub **510** from being displaced axially too far above the counterplate **540**, since this is an axially upward thrust bearing **542** between the shaft end and the counterplate **540**, an opposing bias is typically introduced. This bias is utilized to prevent the thrust bearing gap **542** from becoming too large, which would reduce the effectiveness of the motor **500**. Approaches to this can be seen in the provision of a bias magnet **564** facing the motor magnet **550** and axially spaced therefrom. By selecting a suitable size and location for this bias magnet **564**, an appropriate bias against the shaft **514** being axially displaced too far from the counterplate **540** or the base **560** can be optionally introduced.

[0034] To ensure proper lubrication of the thrust bearing surface **542**, oil pumping grooves **527** are provided. The oil pumping grooves **527** are positioned along a gap **524** between the horizontal body portion **518** of the hub **510** and the sleeve **520**. The oil pumping grooves **527** may be disposed along the surface of either the horizontal body portion **518** of the hub **510**, or the sleeve **520**. Preferably, the grooves **527** are placed along the hub **510**, as shown in **Figure 4** and **5**.

[0035] To inhibit the loss of liquid lubricant from the gaps **522**, **524**, **542**, and **572** during operation, a capillary seal **574** is optionally provided. In conventional arrangements, the capillary seal **574** is located at the end of the sleeve bearing gap **522** distal from the thrust bearing gap **542**. Further information concerning operation of a capillary seal within a bearing gap is disclosed in U.S. Patent 5,524,986 entitled "Fluid Retention Principles for Hydrodynamic Bearings." That patent issued to Seagate Technologies, Inc. in 1996, and is incorporated herein in its entirety by reference. According to the preferred embodiment shown, the capillary seal **574** is located in a third hub-sleeve gap **572** proximate to the horizontal (second) hub-

sleeve gap **524**. The cap seal **574** may also be placed in the gap **524** behind the oil pumping grooves **527**, or not be provided at all.

[0036] Operation of this impeller phenomenon is as follows. When the motor **500** is energized and the shaft **514** and connected hub **510** are rotated, a high pressure region is created in horizontal gap **524** by the oil pumping grooves **527**. This high pressure region impels oil from the gaps **572** and **524** into the sleeve bearing surface **522** and the thrust bearing surface **542**. The lubricating fluid is then impelled into the thrust bearing region **542** to support relative rotation between the bottom end of the shaft **514** and the facing surface of the counterplate **540**. During rotation, the fluid is maintained in the gap **542** by the grooved pattern during rotation. Obviously, when the shaft comes to rest, the shaft end will rest on the plate **540** and, although the volume of fluid is very small, it will tend to be forced back out into the gaps **524** and **572**. Therefore, space must be allowed in these gaps **524** and **572** for this fluid. When the motor **500** is idle, the capillary seal **574** aids in maintaining fluid within the bearing system. Preferably, the gaps **572** and **524** and the volume of lubricating fluid are configured so that the gaps **572** and **524** are dry during rotation of the shaft to prevent any power loss or bearing effect. Preferably, the gaps **522**, **524** and **572** are relatively sized so that if the shaft **514** tilts during rotation, the hub **510** does not contact the sleeve **520** in the gaps **572** and **524**. As noted earlier, in previous designs the capillary seal was located in the bearing gap **522**, proximate to the second gap **524**. The impeller means was located either (or both) in the thrust bearing gap **542**, or in the first journal bearing gap **522**. When the impeller means pushed or pulled oil into the thrust bearing gap **542**, an un-lubricated area would result in the upper portion of the journal bearing **522**. This could lead to damage of the shaft **514** and sleeve **520** resulting from frictional contact in the un-lubricated area, particularly if the shaft **514** were to tilt from vibration. In the present invention, excess oil is now able to be stored in gaps **524** and **572**, with no resulting un-lubricated area in the journal bearing gap **522**. Further, the shaft **514** and sleeve **520** may be shortened as capillary seal is no longer needed along the shaft **514**.

[0037] **Figure 6** depicts a perspective view of the grooved portion of an exemplary hub **510**, illustrating a preferred configuration of the oil pumping grooves **527**. A bottom view is provided for the hub **510**. Any type of pattern as is used to draw oil in a fluid dynamic bearing is suitable to serve as the oil pumping groove.

[0038] **Figure 7** presents a partial cross-sectional view of an improved spindle motor arrangement **600** in an alternative embodiment, in which oil pumping grooves **627** are machined. The motor **600** first comprises a hub **610**. Fitted inside the hub **610** is a sleeve **620**. The sleeve **620** is fit by known means, such as a shrink fit. The hub **610** includes a radial shoulder **612** for receiving and supporting a body to be rotated, such as a magnetic disc (not shown). The hub **610** and sleeve **620** are configured for constant high speed rotation. This rotation is established by a stator **652** which is mounted from a base **660**. The stator defines an electric coil that, when energized, creates a magnetic field. The energized coil cooperates with magnets **650** which are mounted from the inner surface of the hub **610**.

[0039] As noted, the hub **610** and sleeve **620** are configured for high speed rotation. In this respect, the sleeve **620** rotates on a stationary base adapter **640**. **Figure 8** illustrates an enlarged view of the interfaces among the shaft **614**, base adapter **640**, sleeve **620**, and the shield **618**. The interface between the bottom of the sleeve **620** and the top of the base adapter **640** thus defines a thrust bearing **642**. Liquid lubricant is provided along the thrust bearing gap **642** to provide a fluid bearing surface. The top face of the base adapter **640** may optionally include a grooved pattern (not shown) for receiving and holding liquid lubricant when the motor **600** is at rest. When the motor **600** is at rest, the sleeve **620** presses directly on the base adapter **640**. Fluid is then extruded around the inner diameter of the sleeve **620**.

[0040] The motor **600** of **Figure 7** next comprises a shaft **614**. In the arrangement of **Figure 7**, the shaft **614** is stationary. The sleeve **620** is disposed between the stationary shaft **614** and the surrounding base adapter **640** on a lower side and the stationary shaft **614** and the hub **610** on an upper side. Referring to

**Figure 8**, it can be seen that the interface between the stationary shaft **614** and the surrounding sleeve defines an upper second bearing surface **622a** and a lower second bearing surface **622b**. The two second bearing surfaces **622a,b** are separated by a re-circulating hole **680** disposed in the sleeve **620**. The re-circulating hole **680** prevents any pressure difference between the upper **622a** and lower **622b** journal bearings and also allows any air trapped in the lubricating fluid to escape. The second bearing surfaces **622a,b** are substantially vertical, and also receive liquid lubricant. These vertical bearing surfaces **622A,B** can also be configured with grooves. Also notable is a third fluid-holding gap **672** located between a shield **618**, fitted onto the shaft **614**, and the outer diameter of the sleeve **620**. The third gap **672** may also receive fluid.

[0041] To prevent the sleeve **620** and adjoining hub **610** from being displaced axially too far above the base adapter **640**, since this is also an axially upward thrust bearing **642** between the sleeve end and the base adapter **640**, an opposing bias is typically introduced. This bias is utilized to prevent the thrust bearing gap **642** from becoming too large, which would reduce the effectiveness of the motor **600**. Approaches to this can be seen in the provision of a steel bias ring **664** facing the motor magnet **650** and axially spaced therefrom. By selecting a suitable size and location for this bias ring **664**, an appropriate bias against the sleeve **620** being axially displaced too far from the base adapter **640** or the base **660** can be optionally introduced.

[0042] To ensure proper lubrication of the thrust bearing surface **642**, novel oil pumping grooves **627** are provided. The oil pumping grooves **627** are positioned along a horizontal gap **624** between the shield **618** and the sleeve **620**. The oil pumping grooves **627** may be disposed along the surface of either the shield **618** or the sleeve **620**. Preferably, the grooves **627** are placed along the shield **618**, as shown in **Figure 7** and **8**. Alternatively, the large horizontal gap (not numbered) between the base adapter **640** and the sleeve **620**, above the re-circulating hole **680**, may be reduced so that a second set of oil pumping grooves may be disposed

along either the sleeve or the base adapter in the gap. Of course, then the second capillary seal **674b** would be disposed behind the grooves.

[0043] To inhibit the loss of liquid lubricant from the gaps **622a,b**, **624**, **642**, **672**, and **695** during operation, capillary seals **674a,b** are optionally provided. In conventional arrangements, the capillary seal **674a** is located at the end of the sleeve bearing gap **622a** distal from the thrust bearing gap **642**. According to the preferred embodiment shown, the capillary seal **674a** is located in a third hub-sleeve gap **672** proximate to the horizontal (second) hub-sleeve gap **624**. A second capillary seal **674b** is located in a sleeve-base adapter gap **695**. The capillary seal **674a** may also be placed in the gap **624** behind the oil pumping grooves **627**, or not be provided at all. Further, the second capillary **674b** seal may not be necessary.

[0044] The groove pattern **627** is configured so that oil flow is impelled towards the thrust bearing surface **642** when the hub **610** and sleeve **620** are rotated. An example of such a pattern is a spiral pattern machined into the bottom of the shield **618**. Any type of pattern as is used to draw oil in a fluid dynamic bearing is suitable to serve as the oil pumping groove.

[0045] Operation of this impeller phenomenon is as follows. When the motor **600** is energized and the sleeve **620** and adjoining hub **610** are rotated, a high pressure region is created in second gap **624** by the oil pumping grooves **627** impelling oil from the gaps **672** and **624** into the sleeve bearing surface **622** and the thrust bearing surface **642**. The lubricating fluid is then impelled into the thrust bearing region **642** to support relative rotation between the bottom end of the sleeve **620** and the facing surface of the base adapter **640**, the fluid being maintained in the gap **642** by the grooved pattern during rotation. Obviously, when the sleeve comes to rest, the sleeve end will rest on the base adapter **640** and, although the volume of fluid is very small, it will tend to be forced back out into the gaps **624** and **672**. Therefore, space must be allowed in these gaps **624** and **672** for this fluid. When the motor **600** is idle, the capillary seals **674a,b** aid in maintaining fluid within the bearing system. Preferably, the gaps **672** and **624** and the volume of lubricating fluid are

configured so that the gaps **672** and **624** are dry during rotation of the shaft to prevent any power loss or bearing effect. Preferably, the gaps **622**, **624**, **672**, and **695** are relatively sized so that if the sleeve tilts during rotation, the sleeve **620** does not contact either the base adapter **640** or the shield **620**. As noted earlier, in previous designs the capillary seal was located in the bearing gap **622A**, distal from the gap **642**. The impeller means was located either (or both) in the thrust bearing gap **642**, or in the first journal bearing gap **622**. When the impeller means pushed or pulled oil into the thrust bearing gap **642**, an un-lubricated area would result in the upper portion of the journal bearing **622A**. This could lead to damage of the shaft and sleeve resulting from frictional contact in the un-lubricated area, if the shaft were to tilt from vibration. In the present invention, excess oil is now able to be stored in gaps **672** and **624**, and there is no resulting un-lubricated area in the journal bearing gap **622a**. Further, the shaft **614** and sleeve **620** may be shortened as no capillary seal is needed therein.

[0046] **Figure 9** displays the results of a simulation conducted throughout typical taper tolerances in the oil pumping grooves **627** and gaps **624** and **672**, with a zero flow boundary condition imposed at the capillary seal end **674**. The pressure was measured at the capillary seal end **674a**. The results show that the oil pumping grooves **627**, during shaft rotation impel oil towards the thrust bearing surface **642**, resulting in a vacuum effect at the capillary seal end **674a**.

[0047] While the foregoing is directed to embodiments of the present invention, other and further embodiments of the invention may be devised without departing from the basic scope thereof, and the scope thereof is determined by the claims that follow.